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ANNUAL REPORT

CONTRACT NONR - 258 (00)

DEVELOPMENT OF A LIQUID ANNULAR RING TYPE OF AIR COMPRESSOR

To

OFFICE OF NAVAL RESEARCH

Code 429 T 3 - 2716

NAVY DEPARTMENT

Washington 25, D. C.

From

DEPARTMENT OF MECHANICAL ENGINEERING

School of Engineering and Architecture

HOWARD UNIVERSITY

Washington 1, D. C.

Prepared by Darnley E. Howard and Stephen S. Davis

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SUMMARY

This report describes the test procedure for determining the various losses in a liquid annular ring type of gas compressor; the development of a new type of partial bearing (oil); the calibration of a testing motor and the construction and tests of a prototype.

It is shown that this low cost smooth-running medium speed (3000 r.p.m.) compressor in a 3 h.p. size will develop 62 per cent isothermal efficiency and deliver 53 cubic feet of cool oil-free air per minute at a pressure of 4 lbs. per square inch.

INTRODUCTION

This study pertains to a simple type of air compressor in which an annular ring of liquid acting as a sealing and compressing medium (See Figure I) is carried in a casing mounted on anti-friction bearings eccentrically placed relative to a rotor with six to nine peripheral working chambers. On driving the rotor, air is trapped in these chambers, compressed and discharged through suitably placed ports in the tubular rotor support. The casing is caused to be revolved by the friction of the moving liquid at a speed approximately 85 per cent of that of the rotor.

Several patents on devices of this kind have been issued, one of the earliest being No. 1,038,769, granted to Josef Lehne in 1912. Differing from this type in the manner of utilizing the liquid annulus is the Nash Jennings pump originally patented in 1911 (No. 988,133) and now commercially produced by the Nash Engineering Company of South Norwalk, Connecticut. This patent covers a compressor in which the sealing liquid describes an elliptical annulus in a stationary casing.

A thorough search of the available literature has revealed very little experimental data on the performance of the rotating casing type and, since the initial isothermal efficiency of a small model reached 63 per cent, this research program was undertaken with the purpose of making available scientific information on (a) pressure variations due to improper porting of air flow; and (b) the movement of boundary layers of water at various angular positions due to changes of pressure, acceleration of water in the chambers, and relative amounts of water past the rotor.

INITIAL TEST PROCEDURE

Some forty-six tests were made to determine how performance was affected by the following:

1. Variation in eccentricity of rotor relative to casing
2. Variation in casing size with the same size rotor
3. Variation in rotor speed
4. Variation in ambient pressure surrounding rotor and casing
5. Variation in surface smoothness of rotor blades and inside of casing
6. Variation in density and/or viscosity of liquid medium
7. Variation in speed of casing (rotor speed constant)
8. Variation in blade shape (edges and thickness)

As a result of these tests, the possible sources of power loss were classified in the following order, with approximate magnitude indicated:

1. Hydraulic losses (from 55 to 65 per cent of total loss)
   - a. Slipping of water relative to casing
   - b. Reciprocation of blades
   - c. Relative motion between tip of rotor blades and water
   - d. Blade shape
   - e. Flow of water in and out of chambers due to air pressure
   - f. Movement of water due to centrifugal causes (lower angular velocity of casing relative to rotor)
2. Mechanical losses (14 to 20 per cent of total loss)
   - a. Friction of casing bearings
   - b. Friction of rotor bearings
   - c. Friction of casing against atmospheric air

3. Air losses (20 to 30 per cent of total loss)

    a. Friction of air and out of chambers

    b. Acceleration of air in and out of chamber

    c. Leakage of air

4. Miscellaneous

    a. Cutoff - discharge port

    b. Cutoff - inlet port

    c. Re-expansion of clearance air

From the tests mentioned above, the lines of endeavor were indicated. Inasmuch as changes in blade arrangement affected efficiency as much as five (5) per cent and leakage accounted for 7 per cent, it was decided (1) to make a study of blade shapes and concurrently (2) to redesign the rotor and rotor support to reduce clearances and thereby minimize leakage loss.

A detailed study of the relative movement of the boundary layers past various shapes of blades was facilitated by the construction of a model in which two similar blades were mounted at the end of an arm and driven by a small motor. Meanwhile, the casing, complete with a plexiglass cover, was driven by another motor. The transparent cover made possible stroboscopic examination of the phenomonon at all speeds. Power measurement and speed control was made possible by the separate motor drives.

As a result of this study, values were obtained for the coefficient of drag which indicated that the phenomenon lay in the viscous or laminar region. In the actual compressor the rotor is driven and the casing follows, due to hydraulic drag. The blades of one of the original rotors were modified as to number, angle, and shape, and a series of experiments performed. The results are shown in Figure II. It will be noted that the losses due to blade modification were relatively low. But the loss due to plugging the ports was much higher. The reason for this loss probably was because there was a mass flow of water in and out of the pockets. While this loss would not be as great under actual operating conditions, its magnitude indicates the need for further study.

In one test carbon tetrachloride was substituted for the water without any noticeable increase in loss thereby suggesting the use of this denser (specific gravity 1.58) sealing medium with a probable advantage of proportionally higher discharge pressure.

BEARING TEST

Since the leakage loss in the original model amounted to 7 per cent with a radial clearance of 0.0025, it was decided to reduce this clearance. Inasmuch as the rotor was supported as a cantilever, clearance was necessary to allow for deflection of the rotor without touching the support. Furthermore, widening the rotor would increase capacity with no appreciable increase in side cheek loss. To secure these two advantages, an outboard bearing was provided (point A in Figure I).

Since this bearing would be relatively large in diameter (2-1/2"), its power loss due to viscous drag would be so great that a conventional bearing could not be used. After many modifications, a partial bearing of the form shown in Figure III was adopted. It is evident that attempting to test the effect of variations in oils, inlet pressures, radial loads, bearing materials, oil inlet angle and speeds would require an enormous number of runs. So, after about 65 tests, varying the elements above, the following general conclusions were arrived at:

1. A wedge on the partial bearing shoe is necessary for a prompt start of the oil film and reproducible.results.
2. A partial bearing is superior to a conventional bearing.
3. A suitable thin oil (No. 10 SAE) will give best results.
4. The friction coefficient varies downward with an increase in the unit pressure.
5. The friction coefficient decreases with an increase in the size of shoe because of the reduction of side leakage.

Curves were plotted with the generalized dimensionless variable ZN/P as the abscissa, where

Z = viscosity in centipoises

N = speed of journal in rpm

P = psi of projected area

and the coefficient of friction 'f' as the ordinate. It will be seen that this curve (Figure IV) is quite typical and that 'f' decreases with an increase in unit pressure. The coefficient for the partial wedge bearing was 0.019 whereas a conventional bearing gave a coefficient approximately twice as great (f = 0.039).

The instrumentation and test procedure produced results which were reproducible and consistent. Utilizing the probes shown in Figure III, unit pressures could be measured. Since our gage was not calibrated above 600 psi the test stopped at that point although higher pressures would have been obtained with an increase in load.

Having thus determined the direction of best results and having absolute values of load capacity, power loss and coefficient of friction, this series of experiments was brought to an end.

A conventional bearing will be used in the prototype since it is easier to construct and, if necessary, final results can be corrected on the basis of attained efficiency in these experiments.

As a result of the investigation mentioned above, a prototype was constructed incorporating the following features:

1. Reduced clearance (0.001) between rotor and shaft to minimize leakage.
2. Long narrow ports in the rotor for sharper cutoff.
3. Lengthening the lap on the stationary valve (rotor support) to give correct compression before release.
4. Making a relatively long rotor to reduce end plate fluid loss.
5. Adoption of oil sleeve outboard bearing to support the longer rotor.
6. Improved instrumentation: probes were placed at the suction and discharge port to determine pressure changes in these areas.
7. A novel oil return from the sleeve bearing mentioned in 5 above.
8. A plexiglass cover for observation of water line contours in space between casing and rotor.

The assembled pump is quite compact, over-all dimensions being 17" in length, 8" in height, and 7" in width. Nevertheless, at 3290 rpm it delivers 53 cubic feet of air per minute at a pressure of 4 lbs./sq. in.

In order to adequately test the prototype, a 3 HP General Electric shunt motor was secured. Since a V-belt drive was to be used, it was decided that the compressor casing would be used as a prony brake since it could be readily water cooled and the characteristic curve would then automatically include belt loss for each speed. With the addition of a dashpot at the end of the brake arm; a five-pound weight (suspended between rubber tension members); and a spring balance, vibration of the scale pointer was reduced to one division or 1/2 oz. which permitted quite accurate readings of the pull on the brake arm.

TESTS OF THE PROTOTYPE

The first test of the prototype was made on October 6, 1951. Considerable spray escaped from the side of the casing, which seemed to indicate insufficient rotor depth. The efficiency was only 51 per cent. By October 27, 1/4" lucite segments had been added to the rotor, increasing the diameter by 1/2 inch. The eccentricity of rotor to casing was reduced slightly and an efficiency of 54 per cent obtained. These results indicated the need for (1) increasing initial submersion of the blades; and (2) limiting pressures to around 6 lbs./sq. in. which appeared to be the area of maximum efficiency. Accordingly, the lucite segments were removed and the port openings generously filleted. This was accomplished by November 14. These changes reduced the sound of water slapping and it was found that it was better to allow the replacement water to go in with the air as a spray rather than in the passage between the casing and rotor. The capacity was increased to 53 cu. ft./min. and efficiency to 55 per cent.

The quantity of air pumped decreased sharply with an increase in pressure even though leakage in this model was much less than in previous models.

Leakage was determined by charging a 2110 cu. in. tank with air at an absolute pressure of 70 lbs./sq. in. and determining the time required for a drop in pressure to 35 lbs. absolute. The discharge flowed through a pressure regulating valve set for pressures of 1, 3, 5, 7, 9, and 10 lb. gage. These pressures were maintained in the discharge lines and the rotor revolved as in operation. The volumes of air shown in the abscissa of Figure V are the volumes of free air after conversion by use of the following equation:

$$\frac{P_{initial} - P_{final}}{P_{atmospheric}} \times \frac{2110 \text{ cu. in.}}{1728} \times \frac{60}{\text{time in secs.}}$$

At this point an attempt was made to pinpoint the losses and ascribe approximate values as determined from previous isolation tests. It then appeared that the sleeve bearing loss was quite high, and a special test was subsequently made on the bearing. Running it dry, the wattage included motor, belt, and ball bearing loss. This amounted to 163 watts at 3135 rpm. Using kerosene as a lubricant, the wattage was 223 at 3170 rpm and using No. 20 oil, the wattage increased to 456 at 3170 rpm. This indicated that the loss due to the viscosity of the No. 20 oil only was 293 watts. Since it appeared that the high viscous drag cut down on performance, another test was run on January 19, using kerosene as a lubricant. The efficiency was 61.8 percent--a gain of 7 per cent due entirely to the change in oils. (See Figure VI).

Since the diameter of the outboard bearing was but 0.003 greater than the rest of the rotor support tube, the three months of tests, especially under dry and kerosene lubricated conditions, wore it to the point where it was useless and this series of tests was brought to a close.

CONCLUSION

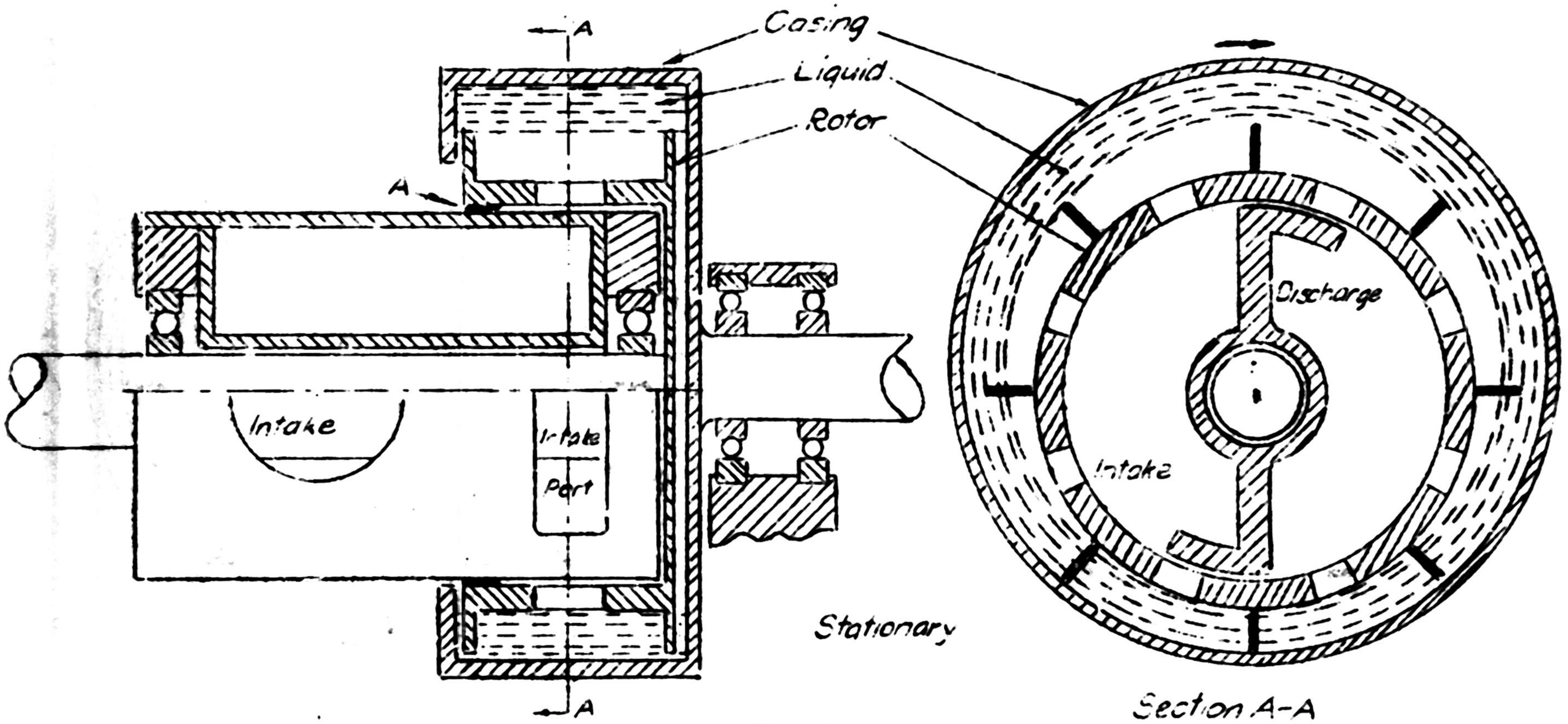
The compression of air in a rotating casing containing a layer of cooling water eccentrically mounted relative to a partitioned rotor is being investigated. An efficiency of 62 per cent was attained and, with further reduction in bearing loss, an increase in efficiency may be achieved. With good efficiency, numerous applications may be found including gas turbine and supersonic wind tunnels.

TABLE 1.- PERFORMANCE DATA.

No. of Run	Pump Speed (rpm)	Amps.	Volts	Dischg. Press. lbs/sq.in.	Nozzle Press. in H_2O	Cu.Ft. Free Air	Motor Watts from Curve	Watts in Dischg. Air	Eff.
1	3100	7.1	224	3.35	14.6	47	910	470	51.7
2	3100	7.3	224	4.02	13.6	44.5	940	557	59.2
3	3100	7.9	223	5.20	11.8	42	1,060	652	61.5
4	3100	7.7	223	5.95	9.0	36.7	1,010	624	61.8

TABLE 2.- DETERMINATION OF LEAKAGE OF FREE AIR THROUGH CLEARANCE PASSAGES AT VARIOUS PRESSURES.

Gage Pressure lbs./sq.in. in Discharge Port	Time (Secs.)	Leakage Free Air (ft.^{3})
10	98.6	1.735
9	113.8	1.510
7	152.5	1.125
5	219.8	.780
3	381.4	.449
1	1,119.2	.153



A
A
A
Casing
Liquid
Rotor
Intake
Intake
Port
Stationary
Discharge
Intake
Section A-A

FIG. I

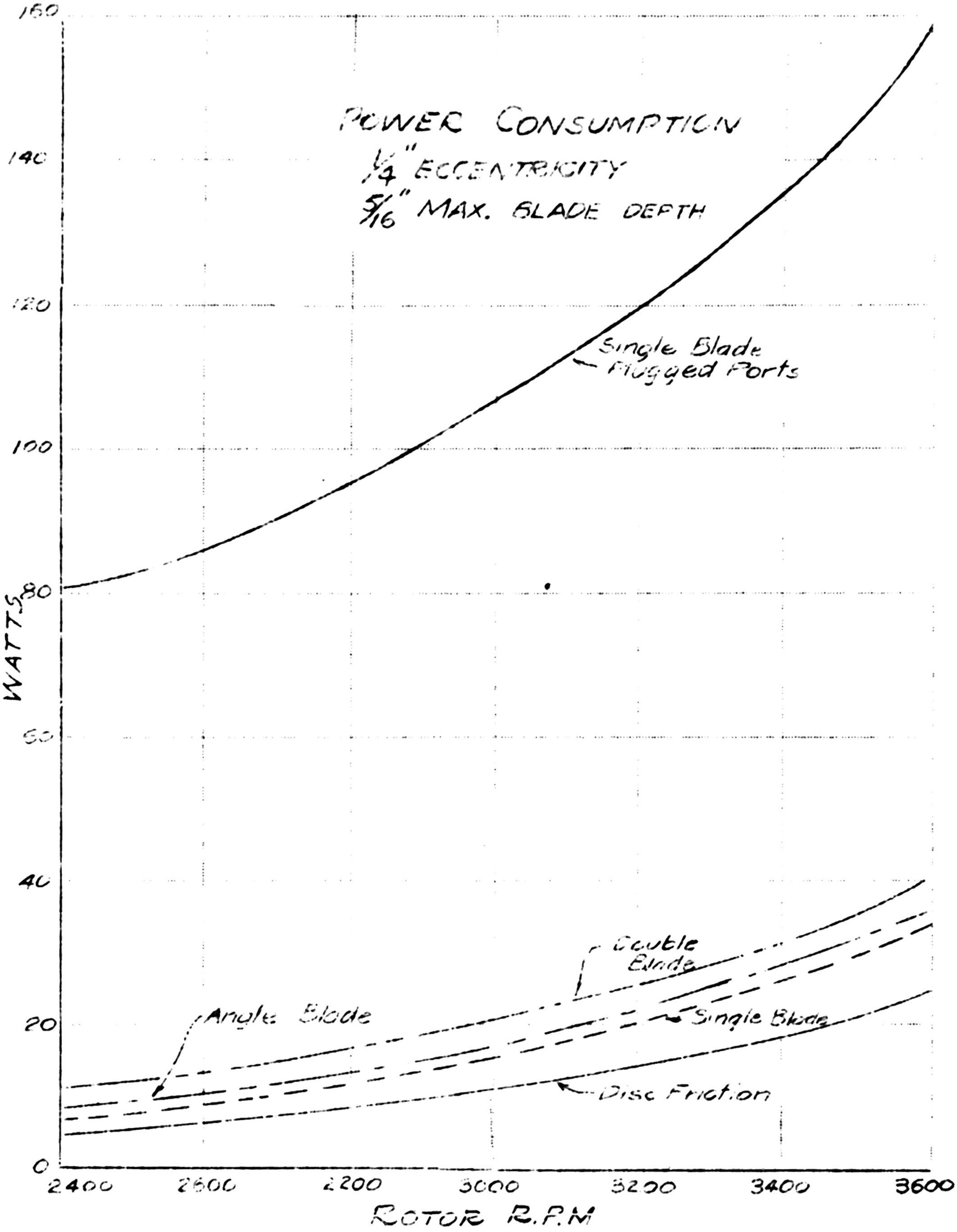
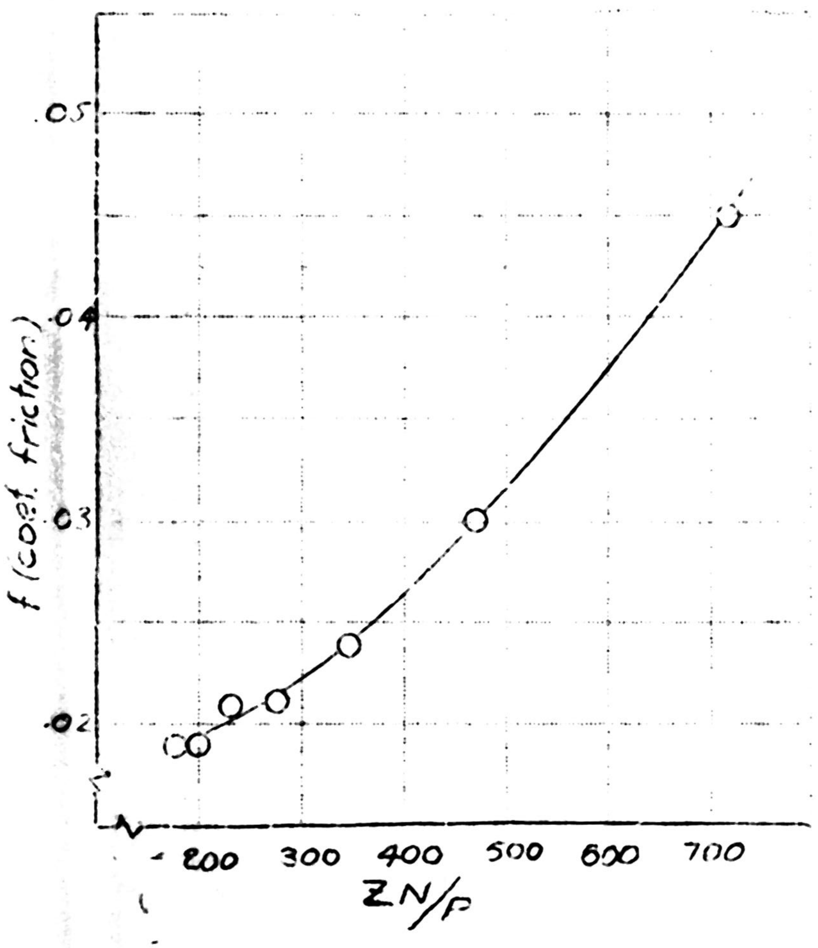
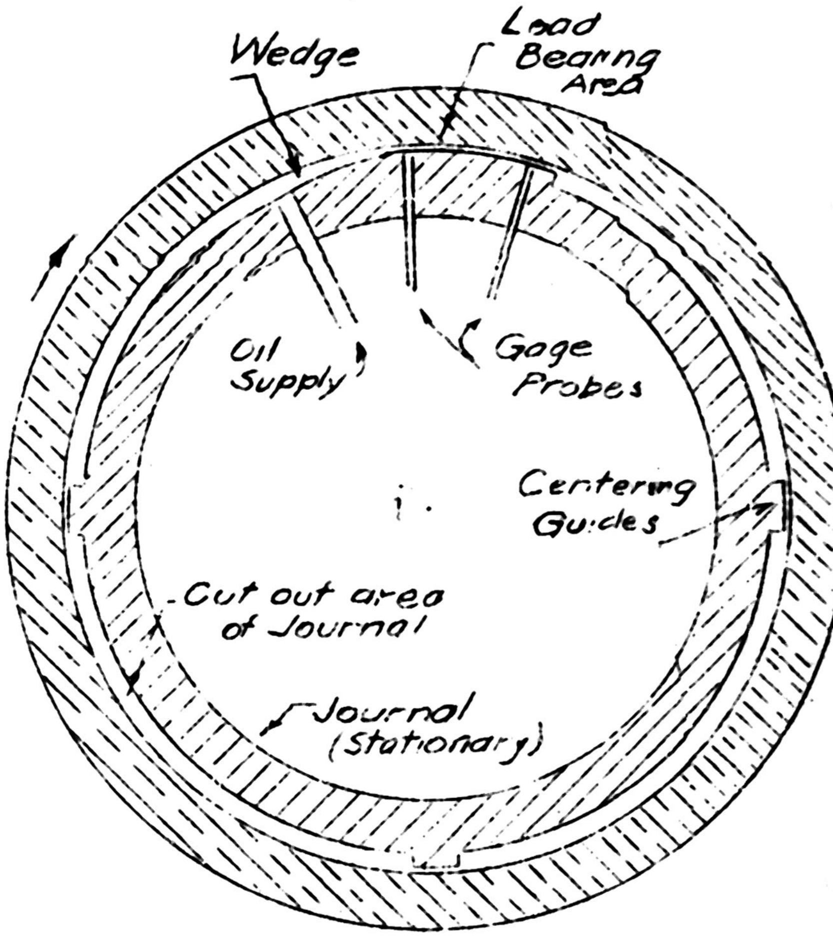

POWER CONSUMPTION
1/4" ECCENTRICITY
5/16" MAX. BLADE DEPTH
Single Blade Plugged Ports
Double Blade
Angle Blade
Single Blade
Disc Friction
WATTS
160
140
120
100
80
60
40
20
0
2400
2600
2800
3000
3200
3400
3600
ROTOR R.P.M

FIG. II

FIG III





PARTIAL BEARING

FIG. IV

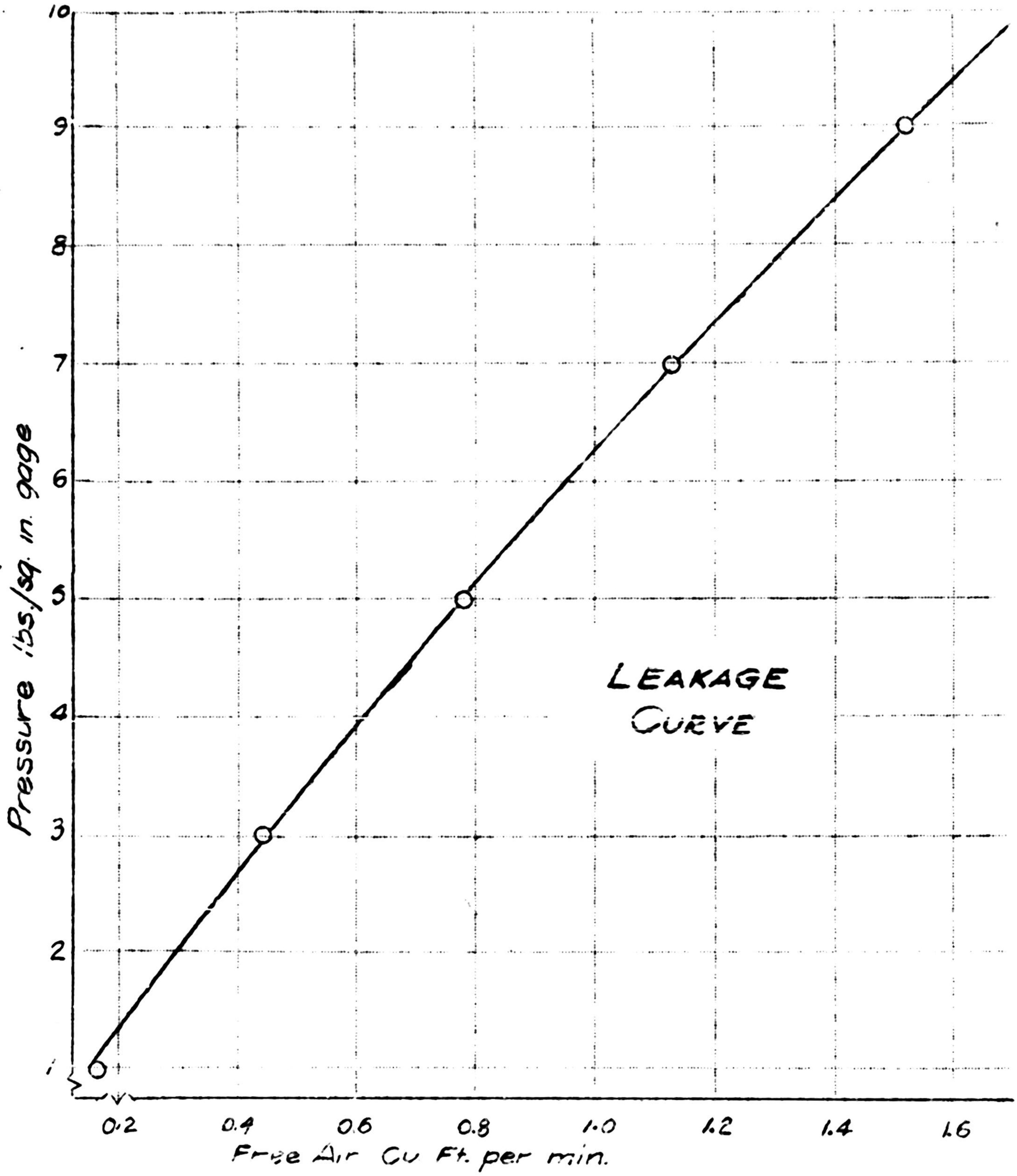
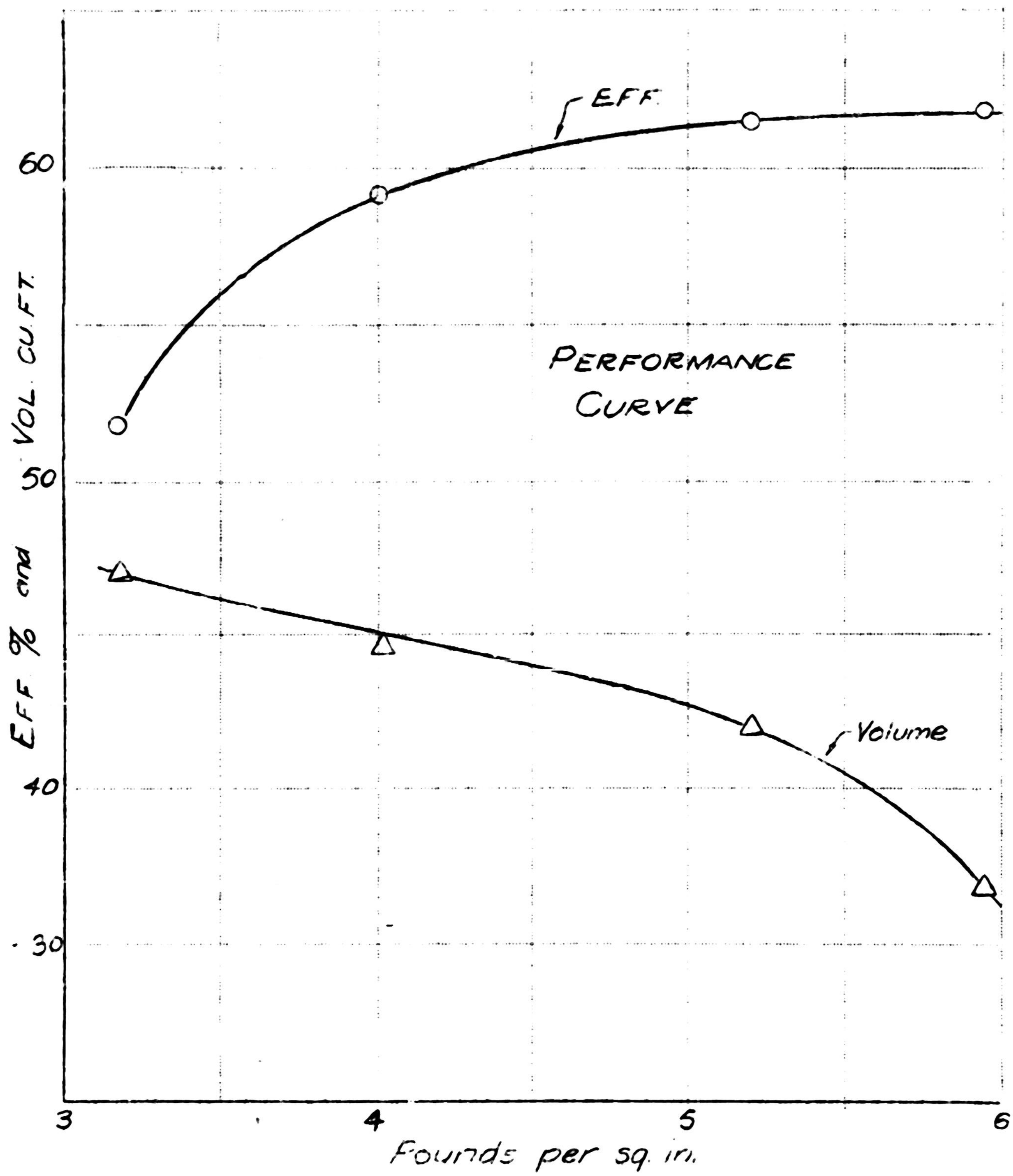

Pressure lbs./sq. in. gage
10
9
8
7
6
5
4
3
2
1
LEAKAGE
CURVE
0.2
0.4
0.6
0.8
1.0
1.2
1.4
1.6
Free Air Cu Ft. per min.

FIG. V



EFF.
PERFORMANCE CURVE
Volume
EFF % and VOL. CU. FT.
60
50
40
30
3
4
5
6
Pounds per sq. in.

FIG. VI

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(Patents)

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1912	1,038,769	Rotary Pump	J. H. Lehne
1912	1,045,732	Turbo-displacement Engine	L. H. Nash
1918	1,262,533	Rotary Compressor	G. C. McFarlane
1918	1,281,972	Rotary Compressor and Exhauster	J. Johnston
1925	1,527,339	Compressor	E. Wilson
1933	1,919,252	Air Compressor	W. W. Paget
1935	2,006,366	Rotary Compressor	W. W. Paget